DaimlerChrysler AG

Method for operating an internal combustion engine

5 The invention relates to a method for operating an internal combustion engine as claimed in claim 1.

By way of example, DE 195 19 663 A1 has disclosed a method for operating an internal combustion engine with
10 spontaneous ignition, in which in a first stage a homogenous, precompressed fuel/air mix which is not suitable for spontaneous ignition is provided in the working space, and in a second stage an additional quantity of the same fuel is injected into the working
15 space in order to bring about the spontaneous ignition. In this case, the fuel/air mix is prepared by means of external mix formation and introduced into the working space, where it is compressed to close to the spontaneous ignition point. The injection of the
20 additional quantity of fuel in the second stage takes place in finely atomized form, avoiding contact with the walls, so as to form a mix cloud in which, on the one hand, the fuel/air ratio is no greater than the stoichiometric mixing ratio and in which, on the other
25 hand, the spontaneous ignition condition is achieved.

Furthermore, DE 198 52 552 C2 has disclosed a method for operating a four-stroke internal combustion engine which, at part-load, forms a lean base mix from air,
30 fuel and retained exhaust gas and, at full load, forms a stoichiometric mix. At part load, compression ignition takes place, whereas at full load spark ignition takes place. Furthermore, mechanically controlled exhaust-gas retention with switchable valve
35 closure overlap and exhaust gas throttling is provided. An activation injection can be carried out into the retained exhaust gas. The quantity of retained exhaust gas, with the valve closure overlap switched on, is

controlled or preset as a function of the engine speed and engine load by an exhaust-gas throttle valve which is active for all the combustion chambers. The pressure when the intake members open into the individual
5   combustion chambers is evened out by a cylinder-selective, cylinder-consistent activation injection.

A method for operating a four-stroke, reciprocating-piston internal combustion engine is also known from
10 DE 198 18 569 C2. It is characterized by a homogenous, lean base mix of air, fuel and retained exhaust gas and by compression ignition and direct injection of the fuel into the combustion chamber. The volume of the combustion chamber changes cyclically. The combustion
15 chamber can be filled with fresh gas through at least one intake member, while the combustion exhaust gases can be at least partially expelled through at least one exhaust member. In the part-load range and in the lower full-load range, the internal combustion engine is
20 operated with compression ignition and preferably mechanically controlled exhaust-gas retention, whereas in the full-load range and high part-load range it is operated on the Otto cycle.

25 One drawback of the methods known from the above-mentioned documents is in particular that during the compression ignition of homogenous mixes, the start of combustion and the release of heat cannot be controlled. The control of the exhaust-gas quantity by
30 exhaust-gas retention and exhaust-gas suck-back are expensive and difficult to realize.

By contrast, it is an object of the invention to provide a method for operating an internal combustion
35 engine in which the reaction state of the mix can be influenced in a targeted way.

This object is achieved by the method having the features of claim 1.

The method according to the invention is distinguished

5   by the fact that the mass of fuel to be injected is divided into a preinjection and a main injection, the preinjection taking place into the intermediate compression stroke and the main injection taking place synchronously with the induction. This type of division

10   of the injected fuel masses allows the combustion position to be influenced in a targeted way, since this preinjection into the intermediate compression stroke can be used to influence the temperature and composition of the working gas and therefore the

15   reaction state of the mix.

In a further configuration of the invention, the mass ratio of the fuel mass injected in the preinjection and the fuel mass injected in the main injection is divided

20   according to the operating state of the internal combustion engine. In a preferred embodiment of the invention, the division of the injected fuel masses into preinjection and main injection is approximately 50:50.

25

In a further configuration of the invention, the valve closure overlap between the intake and exhaust valves remains constant during load changes.

30   Furthermore, it is advantageous if the injection point of the preinjection is dependent on the rotor speed and the injection pressure.

Further features and combinations of features will

35   emerge from the following description with reference to the figures, in which:

Fig. 1   shows a graph explaining the division of the
injection quantity between preinjection and
induction-synchronous main injection;

5 Fig. 2   shows a graph illustrating the combustion
position as a function of the injection
quantity division and the air/fuel ratio;

Fig. 3   shows a graph illustrating the charge exchange
10   mean effective pressure as a function of the
injection quantity division and the air/fuel
ratio; and

Fig. 4   shows the pressure, temperature and heating
15   curve in the ignition and gas exchange TDC as a
function of the air/fuel ratio.

The driving factor behind research and development in
internal combustion engines is the desire to constantly
20 improve fuel consumption while, at the same time,
reducing untreated emission levels. In the case of
externally ignited internal combustion engines, in
particular alternative load control methods are
recommended to increase the part-load efficiency. The
25 most important development trends are the stratified
direct injector, which with the aid of quality control
moves the externally ignited internal combustion
engines closer to the spontaneously igniting internal
combustion engines (diesel engine), and the variable
30 valve gear combined with residual gas strategies, which
are intended to limit charge exchange losses. Both
methods theoretically promise major benefits but are
thwarted in one case by the expensive aftertreatment of
the exhaust gas from the superstoichiometric mix and in
35 the other case by the limited residual gas
compatibility of externally ignited internal combustion
engines. The ideal is a link between these two methods:
a quality-controlled internal combustion engine with

high residual gas content and spontaneous ignition,
which on account of homogenous combustion in
superstoichiometric operation emits very little if any
nitrogen oxide.

One property of homogeneous combustion methods is the
spontaneous ignition time, which is determined by the
temperature or mix composition. If the required
charging temperatures are realized with the aid of
exhaust-gas retention, more specifically by means of
the parameters exhaust-gas temperature and quantity,
the result is that the combustion position of cycle n
is dependent on the preceding cycle (n-1); the required
spontaneous ignition temperature is not reached in
extreme circumstances. The combustion position for its
part is the determining factor for the target variables
of the internal combustion engine and therefore must
have values which are defined as a function of load and
engine speed.

It is an object of the invention to find ways of
implementing the changes in exhaust-gas quantity and
temperature which are required in the event of a change
in operating point within the part-load range covered
by the chamber ignition combustion without adversely
affecting the combustion.

Exhaust-gas retention can in principle be achieved with
the aid of suitable control times. This requires
firstly early closing of the exhaust valve, in order to
keep the required quantity of residual exhaust gas in
the combustion chamber of the internal combustion
engine. To prevent the hot exhaust gas from flowing
back into the induction pipe, with ensuing cooling
effects and charge losses, at the same time the intake
valve is opened later. However, this concept cannot be
applied to conventional, externally ignited internal
combustion engines without further measures.

If this valve closure overlap is made sufficiently variable, the first control concept results for this form of providing the required temperature. The
5   requirement for an unaffected high-pressure part and therefore optimum charging in this case, however, presupposes the use of a fully variable valve gear, in which opening time and closing time can be adjusted independently of one another.
10

Returning to the conventional camshaft, the setting of a defined exhaust-gas retention rate remains the job of the camshaft actuators which are already in widespread series use. As an undesirable side-effect, with a rigid
15   cam contour, the angle at which the valve opens changes with the angle at which the valve closes, which leads to charging and efficiency losses and not least to a restricted operating range in terms of load and engine speed.
20

In addition to the control of the temperature at the end of compression with the aid of the exhaust-gas retention rate or quantity, the use of the direct injection and the operation of the internal combustion
25   engine with excess air also influences the working gas temperature and/or the mix composition of the fuel. The effect of the direct injection can in this case be divided into two mechanisms: firstly, a thermal effect, which provides an increase in the exhaust-gas
30   temperature as a result of the conversion of the pre-injected fuel, and secondly the occurrence of a pre-conditioning of the fuel, which increases the reactivity of the latter and therefore influences the integral ignition delay.
35

With the restricted variability and dynamics of the mechanical valve gear with a fixed cam contour, the injection point or the division of the injection

quantity between various injection points becomes the
most important parameter in the control of the
combustion position. Fig. 1 illustrates the division of
the injection quantity between a preinjection and an
induction-synchronous main injection by way of example.

The valve closure overlap in this case allows an
injection prior to the gas exchange TDC. The main fuel
quantity continues to be introduced synchronously with
the induction.

Fig. 2 shows the way in which the combustion position
is dependent on the mentioned injection division. The
test was carried out at a reference point (2000 rpm,
approx. 3 bar p_{mi}) for various valve closure overlaps,
which were kept constant over each variation. In this
context, dashed lines denote constant exhaust control
times. If one of these lines is followed, the result,
in the optimum situation, is a shift in the combustion
position by over 10° crank angle simply by changing the
division. Moreover, it will be noticed that with a
constant intake and exhaust phase position, higher
air/fuel ratios are reached with a smaller preinjected
fuel quantity.

The changes in air/fuel ratio result from a change in
the intake fresh air mass, since the total injection
quantity is kept constant. The shift in the combustion
position is therefore closely associated with the
operations which take place in the charge exchange
loop. A glance at the charge exchange mean effective
pressure in fig. 3, measured from BDC to BDC, shows
that given an approximately even division in the
injection quantity between preinjection and main
injection and high air/fuel ratios, positive values are
obtained for the charge exchange work, even though the
latter ought actually to be negative on account of
induction, exhaust and wall heat losses. The injection

of the fuel into the hot exhaust gas, which at the gas exchange TDC has a sufficiently high temperature level for spontaneous ignition and moreover is equipped with residual oxygen on account of the superstiochiometric
5   operation, allows conversion to be assumed.

Conclusions as to the mechanisms of this conversion can be taken from fig. 4. Starting from the point with the maximum charge exchange work from the engine map shown
10   above with a division of 50:50 and $\lambda \approx 1.4$, the air/fuel ratio λ was increased by supercharging.

First of all, in the non-supercharged state, the conversion can be recognized in the asymmetric pressure
15   and temperature curve. Both maxima lie after the top dead center. If a heating law at the GTDC is formed using mass calculations according to fundamental thermodynamic equations, a release of heat will be revealed. If the degree of charging and therefore the
20   residual air content in the exhaust gas is now increased in steps, the degree of conversion at the charge exchange TDC rises despite otherwise constant boundary conditions, which implies a lack of oxygen. Despite the excess air generated in this way, not all
25   of the preinjected fuel mass can be converted. The preinjection takes place approximately 60° CA before the gas exchange TDC, i.e. in this case mix formation problems and even the formation of carbon particulates have to be assumed. However, these carbon particulates
30   are oxidized again in the main conversion. An improvement in the degree of conversion in the intermediate compression leads to an increase in the temperature and pressure of the retained exhaust gas. The increase in pressure even when the intake valve is
35   opened also explains the drop in the air/fuel ratio with large preinjection quantities in the situation shown above. This higher temperature level which is now required leads to an early shift in the combustion.

However, since a proportion of the fuel, which becomes ever greater according to the degree of charging, is already being burnt at the GTDC, the main conversion is always lower. The exhaust-gas temperature therefore drops and in turn influences the conversion in the intermediate compression.